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INVESTIGATION OF TWIN-CYLINDER IC ENGINE VIBRATION

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***Abstract* — This paper presents the simulation and analysis of the dynamic of a twin-cylinder internal combustion engine, with focus on balancing the shaking forces and moments in order to reduce vibrations. Low vibrations are critical to engines longevity and usability. Therefore, the work discussed herein uses nine different engine configurations to reduce first order vibrations. These are vibrations purely due to the reciprocating masses of the engine. The second order vibrations due to the kinematics of the slider crank mechanism are studied and controlled using five geometry changes. First, we briefly discuss the fundamentals of internal combustion engine vibrations in the context of shaking forces and moments, then we use Virtual Engine software package from FEV to simulate and analyse a twin-cylinder engine. The proposed simulations allow us to analyse the influence of engine configuration as well as effects of changing components such as piston mass on the vibration characteristics of the engine. The results show how the addition of balance shafts can greatly lower engine vibrations or how choosing a configuration to fit the application can further effect vibration of engines such as a 90° Vee twin with naturally low vibration. Additionally, we show in our simulation the effect of low mass crank, conrod, and piston on shaking forces and moments.**

Keywords: Engine, Vibration, twin-cylinder, shaking forces, shaking moments.

I. INTRODUCTION

Most mechanical systems that are subject to time-varying disturbances encounter vibrations. In order to design and build mechanical equipment, one must first predict and control those disturbances. The most common variables measured on a vibrating structure are mode shapes, accelerations, and displacements. These parameters are very important in building a machine that is comfortable for the operator. This is no different for internal combustion engines. In fact, an internal combustion engine can be the best opportunity to reduce or heighten vibrations.

Internal combustion engines are found in the majority of items that allow us to enjoy life [1]. They are found in everything from our automobiles to our lawn care equipment. Some of the most common engines used today are of the single or twin cylinder configuration. These small engines are often placed in smaller equipment or vehicles and have the potential to be used by an operator for hours at a time. The majority of these internal combustion engines are of the reciprocating piston type. One particular type that we will focus on in this work is the twin cylinder engine.

Some of the most common places for a twin cylinder engine configuration are in motorcycles, snowmobiles, all-terrain vehicles, mowers, etc. This is due to the overall packaging, performance characteristics and maintainability of these engines. Engine of the reciprocating piston type produce a lot of vibration [2]. To eliminate as much vibration as possible, a balancing system is often needed [3]. If the engine is not balanced internally, the forces will pass from the piston through the connecting rod and into the crankshaft. At that point, they will transfer through the main bearings and onto the crankcase. From the crankcase they pass into the frame through the engine mounts and lastly to the rider in the case of motorcycle application. The rider can then feel these forces as an annoyance or even as severe as an inability to ride the product.

Having a smooth-running engine isn't the only factor in creating a happy owner. The owner of the vehicle also wants longevity and performance out of their engine. A balanced engine, with less shaking forces and moments will result in less wear on the internal components therefore, in this paper, we will learn about engine unbalance, what causes it and how we can cope with it, as well as study various twin cylinder engine configurations and how each effects vibration.

II. ENGINE VIBRATIONS

A smooth-running engine has multiple advantages ranging from comfort and longevity to higher RPM operation and thus increase power output. A smooth-running engine is also less noisy and more efficient [4]. However, vibrations and harshness are inherent to internal combustion engines due to their intermittent combustion process and internal configuration [5]. These vibrations can be classified as 1/2 order, 1st order, and 2nd order [6]. In general, the designation of order is used to track the relationship between response (vibration and/or noise) created by a rotating component and the rotation RPM. So, for example a 4-stroke single cylinder engine fires once every two revolutions of the crank, thus the vibration created by the combustion is termed a 1/2 order vibration. Another type of vibration in an engine is from the reciprocating masses. These forces shake the engine once every crank revolution and therefore are referred to as primary or first order vibrations [7]. For engines, the whole order definition breaks down when discussing second order vibrations because they are supposed to refer to shaking forces or moments occurring twice in one crank rotation. But second order vibrations are due to kinematic configuration of the slider-crank mechanism. The piston must travel a longer distance in one of the halves of the crank rotation than on the other half. Which will result in an alternation of faster and slower speed. Consequently, unequal inertia forces are created with the accelerations and decelerations of the piston in each of the halves of its path [8] resulting in vibrations termed second order vibrations.

Commonly, only the first and second order vibrations are discussed, and manufacturers effort has been to control these vibrations internally with reductions in rotating and reciprocating masses, engine configurations and balance masses and shafts [9]. Externally, vibration isolating engine mounts are used [10]. These processes are discussed elsewhere in literatures [11]. As previously stated, our objective in this paper is to investigate first and second order vibrations in a twin-cylinder engine and explore the effect of light components and balance shafts on their vibration characteristics. We also have considered the effect of engine configurations on shaking forces and moments.

III. ENGINE MODELLING

The engine is modelled using Virtual Engine by FEV [12]. The software allows the user to create virtual prototypes of engine subsystems and analyse them like you would a physical prototype. Using Virtual Engine, you can quickly create virtual models of crank trains, as well as valve trains, accessory drives, etc. due to its modular modelling approach and then analyse the models to understand their performance and behaviour. The main benefit of using FEV Virtual Engine is that it enables you to work faster and smarter by letting you have more time to study and understand how design changes affect performance. The software allows user to explore the performance of engine designs and refine the design before building and testing a physical prototype.

Virtual Engine provides a modular modelling approach allowing the analysis of subsystems. Each subsystem is derived from a template, which acts as a blueprint and defines the topology. Subsystems can then be adjusted and refined to varying fidelities as appropriate for the analysis. Being we will be focusing our engine balance analysis on a two-cylinder gasoline internal combustion engine; we can build a template using generic dimensions. This generic template is then brought into a subsystem to input specific dimensions and properties from an initial design. Most analysis are completed after the first set of CAD is finished. This gives the closest geometry of what the designer is hoping for. At that point, the inertia and mass properties are taken from CAD and used to build a model in Adams. We can then adjust the balance of the crank and balance shaft to get a well-balanced engine designed for the given situation. For the balance study that is conducted in this paper, we began by building a generic Vee twin engine with three balance shafts. The assembly is shown in Fig. 1.

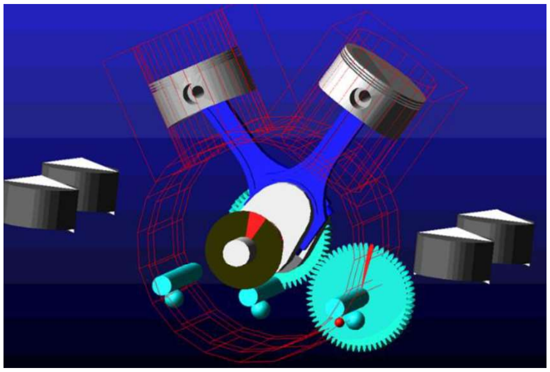
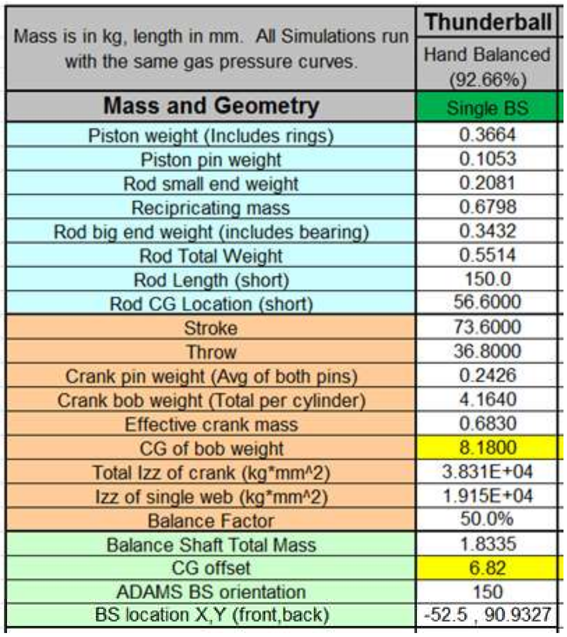


Fig. 1. FEV VE Vee-twin model

This model is initially built in the generic fashion. It is attached to a dyno at the crank and has a fuel map for combustion pressure to power the engine. The balance shafts can be disabled and enabled as needed for the work presented here. The first study that is going to be performed will be using the inertial properties of a known motorcycle engine. The crankshaft is hand balanced to 50% thus a balanced shaft with a 50% balance factor is also built into the model. This will give us a baseline on engine vibration that a rider finds acceptable and enjoyable.

The baseline engine data is shown in Table 1.

Table 1 Baseline Engine Simulation Data



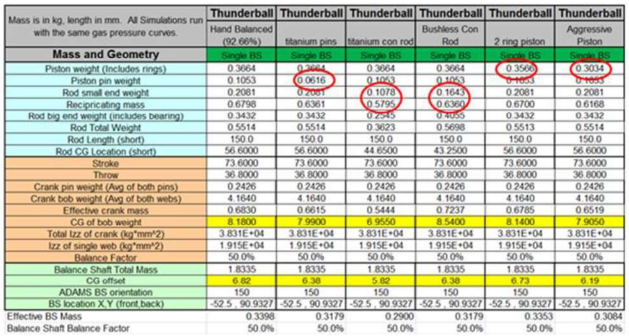
Mass is in kg, length in mm. All Simulations run with the same gas pressure curves.	Thunderball Hand Balanced (92.66%)
Mass and Geometry	Single BS
Piston weight (Includes rings)	0.3664
Piston pin weight	0.1053
Rod small end weight	0.2081
Recipricating mass	0.6798
Rod big end weight (includes bearing)	0.3432
Rod Total Weight	0.5514
Rod Length (short)	150.0
Rod CG Location (short)	56.6000
Stroke	73.6000
Throw	36.8000
Crank pin weight (Avg of both pins)	0.2426
Crank bob weight (Total per cylinder)	4.1640
Effective crank mass	0.6830
CG of bob weight	8.1800
Total Izz of crank (kg*mm^2)	3.831E+04
Izz of single web (kg*mm^2)	1.915E+04
Balance Factor	50.0%
Balance Shaft Total Mass	1.8335
CG offset	6.82
ADAMS BS orientation	150
BS location X,Y (front,back)	-52.5 , 90.9327

With the baseline model build, different simulations are performed to study what could increase or decrease the shaking forces on the engine. First, individual components are modified in order to reduce 2nd order vibrations and then different engine layouts are considered to alter 1st order shaking frequencies.

IV. REDUCING SECOND ORDER VIBRATIONS

As previously discussed, second order vibration results from accelerations and decelerations of the reciprocating masses. These forces are caused by the way the crank, rod, and piston interact. They are higher frequency events and the type that make motorcycle rider hands numb. Using light weight pistons and rods can reduce second order vibrations. Five different changes are made to alter the balance of the Virtual Engine model. The results can be used in deciding whether using parts like titanium connecting rods is worth the extra cost of manufacturing. The five studies performed in this section are: titanium pins, 2 ring piston (changed from typical 3 ring piston), aggressive piston weight reduction, titanium connecting rods, and brushless connecting rod. The parameters for each study are shown in Table 2.

Table 2 Parameters for Component Study



Mass is in kg, length in mm. All Simulations run with the same gas pressure curves.	Thunderball Hand Balanced (92.66%)	Thunderball titanium pins	Thunderball titanium con rod	Thunderball Bushless Con Rod	Thunderball 2 ring piston	Thunderball Aggressive Piston
Mass and Geometry	Single BS	Single BS	Single BS	Single BS	Single BS	Single BS
Piston weight (Includes rings)	0.3664	0.3664	0.3664	0.3664	0.3566	0.3034
Piston pin weight	0.1053	0.0616	0.1053	0.1053	0.1053	0.1053
Rod small end weight	0.2081	0.2081	0.1078	0.1643	0.2081	0.2081
Recipricating mass	0.6798	0.6361	0.5795	0.6360	0.6700	0.6168
Rod big end weight (includes bearing)	0.3432	0.3432	0.2545	0.4055	0.3432	0.3432
Rod Total Weight	0.5514	0.5514	0.3623	0.5698	0.5513	0.5514
Rod Length (short)	150.0	150.0	150.0	150.0	150.0	150.0
Rod CG Location (short)	56.6000	56.6000	44.6500	43.2500	56.6000	56.6000
Stroke	73.6000	73.6000	73.6000	73.6000	73.6000	73.6000
Throw	36.8000	36.8000	36.8000	36.8000	36.8000	36.8000
Crank pin weight (Avg of both pins)	0.2426	0.2426	0.2426	0.2426	0.2426	0.2426
Crank bob weight (Avg of both webs)	4.1640	4.1640	4.1640	4.1640	4.1640	4.1640
Effective crank mass	0.6830	0.6615	0.5444	0.7237	0.6785	0.6519
CG of bob weight	8.1800	7.9900	6.9550	8.5400	8.1400	7.9050
Total Izz of crank (kg*mm^2)	3.831E+04	3.831E+04	3.831E+04	3.831E+04	3.831E+04	3.831E+04
Izz of single web (kg*mm^2)	1.915E+04	1.915E+04	1.915E+04	1.915E+04	1.915E+04	1.915E+04
Balance Factor	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%
Balance Shaft Total Mass	1.8335	1.8335	1.8335	1.8335	1.8335	1.8335
CG offset	6.82	6.38	5.82	6.38	6.73	6.19
ADAMS BS orientation	150	150	150	150	150	150
BS location X,Y (front,back)	-52.5 , 90.9327	-52.5 , 90.9327	-52.5 , 90.9327	-52.5 , 90.9327	-52.5 , 90.9327	-52.5 , 90.9327
Effective BS Mass	0.3398	0.3179	0.2900	0.3179	0.3353	0.3084
Balance Shaft Balance Factor	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%

Being the reciprocating mass is changing in each study, the crankshaft has to be rebalanced to eliminate any primary forces. All crankshafts are kept at a 50% balance factor, as well as the balance shaft. Each configuration will be run at a cruising speed of 2931 rpm as well as a wide-open throttle speed of 9000 rpm. The shaking forces and moments for each engine are analysed and discussed. The results are shown in Tables 3 and 4 and Figs. 2 and 3.

Table 3. 2931 RPM Component Results

2931 RPM Peak Results	Std Tball	Balanced Light Pin	Bal Tit Con Rod	Bal Bushless Con Rod	Bal 2 ring piston	Balanced Aggressive Piston
Vertical Force Pk-Pk (N)	1,058	995	888	995	1,046	966
Horizontal Force Pk-Pk (N)	1,038	968	880	968	1,024	942
Yaw Moment Pk-Pk (N*m)	58	54	45	58	57	53
Pitch Moment Pk-Pk (N*m)	57	54	46	59	57	53
Roll Moment Pk-Pk (N*m)	932	928	1,000	948	932	925

Fig. 2. RPM Component Results

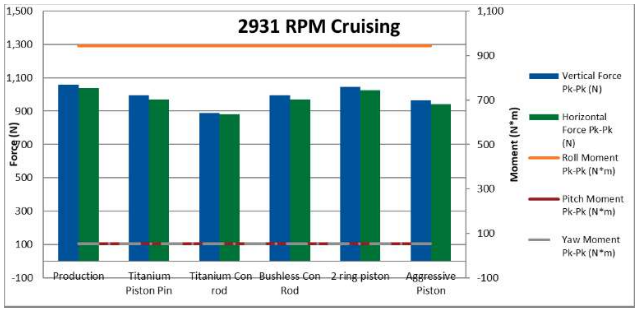


Table 4. 9000 RPM Component Results

9000 RPM Peak Results	Std Tball	Balanced Light Pin	Balanced Tit Con Rod	Balanced Bushless ConRod	Balanced 2 Ring Piston	Bala Aggr Pis
Vertical Force Pk-Pk (N)	9,974	9,380	8,370	9,387	9,861	9,1
Horizontal Force Pk-Pk (N)	9,794	9,142	8,306	9,134	9,656	8,8
Yaw Moment Pk-Pk (N*m)	542	511	420	550	536	49
Pitch Moment Pk-Pk (N*m)	539	510	432	556	533	49
Roll Moment Pk-Pk (N*m)	3,163	2,764	3,437	3,099	3,128	2,9

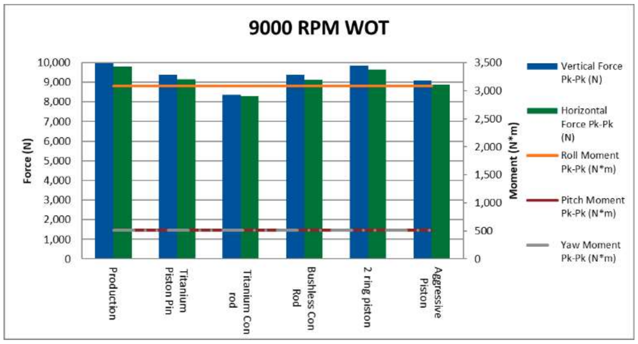


Fig. 3. 9000 RPM Component Results

From the results in Table 3 and 4 and Figs. 2 and 3, it can be seen how changing the masses of key components (piston, connecting rod, piston pin) affect the shaking forces and moments. The cost of the different piston, pin or connecting rod would have to be weighed with the benefits seen in the graphs. The changes are not drastic and may not be noticed by the operator. Depending on the end goal, this slight decrease could be effective.

The titanium connecting rod reduces the shaking forces as well as the pitch and yaw moments, but in turn it increases the roll moment slightly. On the other hand, reducing the reciprocating mass of the system by using a titanium piston pin or an aggressive piston mass reduction can reduce all the forces and moments when comparing to the baseline results

V. REDUCING FIRST ORDER VIBRATIONS

In this section of the simulation, we are going to study the effects of different engine configurations. These are going to include: adding a second 1st order balance shaft, adding a 2nd order balance shaft, a split pin crankshaft, a 90-degree V-twin, parallel twin in both 180- and 270-degree firing orientation, and lastly a boxer configuration.

A. *60 DEGREE V-TWIN ENGINE SINGLE 1ST ORDER BALANCE SHAFT*

This is the most common configuration that is used by many of the top motorcycle companies. This is also the baseline used herein. In Fig. 4, we can see how the model is setup with the dual balance shafts. The second shaft is placed at the front of the engine the same distance as the baseline. The balance shaft is reduced to 25% balance factor. This should reduce some moment.

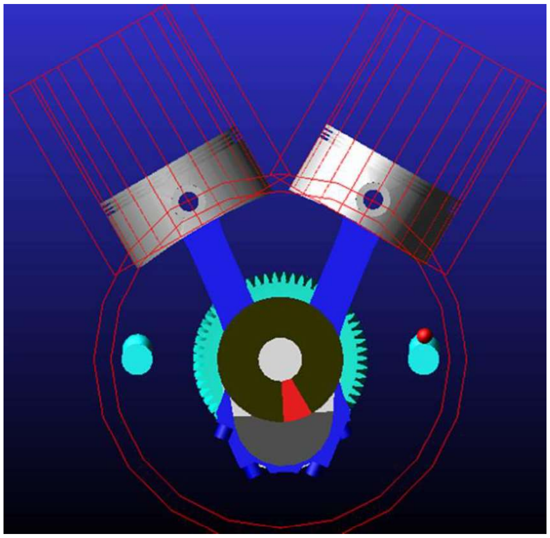


Fig. 4. 60 Degree V-Twin; Dual Balance Shaft Setup

B. *60 DEGREE V-TWIN ENGINE WITH A 1ST ORDER BS AND A 2ND ORDER BS*

Figure 5 shows the setup for an engine with a 2nd order balance shaft added. The baseline eliminated primary forces and now this added balance shaft will eliminate the secondary forces. This is done by spinning the shaft in the same direction as the crankshaft at twice the velocity. This would also make a more complicated design to produce, but depending on the application it might be necessary.

C. *60 DEGREE V-TWIN ENGINE WITH A SPLIT CRANK PIN*

In this split crank analysis, we will look at two different iterations. The first is a 90 degree split and the second is a 30 degree split. The 30 degree split will mimic a 90 degree Vee due to the 60 degree cylinder angle. This should be the better of the two designs and we will review it in the results section by looking at the models built in Fig. 6, we can see how the 30 degree split crank looks like it will perform better. The connecting rods are operating at close to opposite angles similar to a 90 Vee.

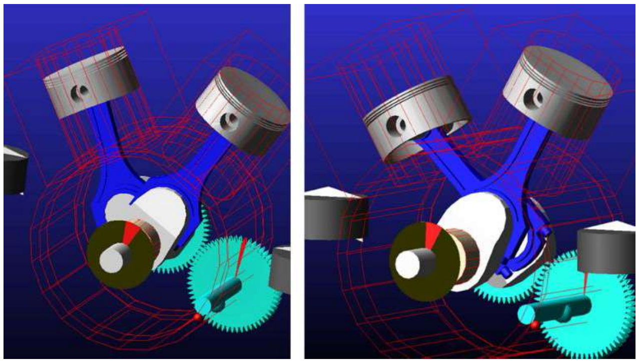


Fig. 5. 60 Degree V-Twin; 2nd Order Balance Shaft

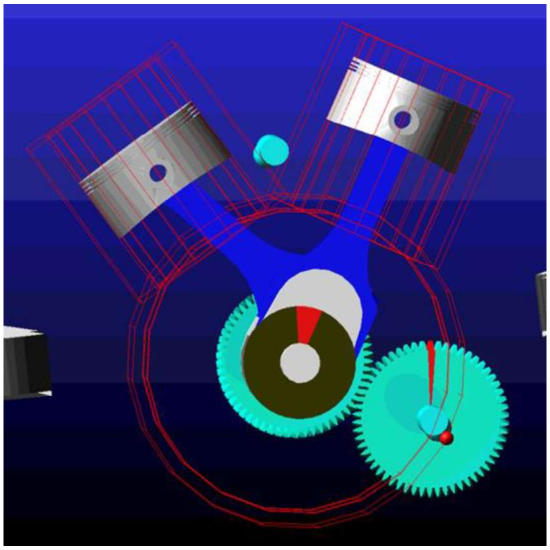


Fig. 6. Split Crank. Left: 90° split. Right: 30° split.

D. *90 DEGREE V-TWIN ENGINE*

This is the configuration that was discussed earlier. One of the simplest engine configurations there is to model as well as manufacture and seen on Fg.7.

E. *PARALLEL TWIN: 180 DEGREE AND 270 DEGREE FIRING*

The parallel twin will be analysed in its two most common configurations. The 180 degree and 270 degree firing offset. This translates to 180 and 90-degree crank pin respectively. Both setups have a balance shaft that spins opposite the crank in order to eliminate primary forces. The two configurations can be seen in Fig. 8.

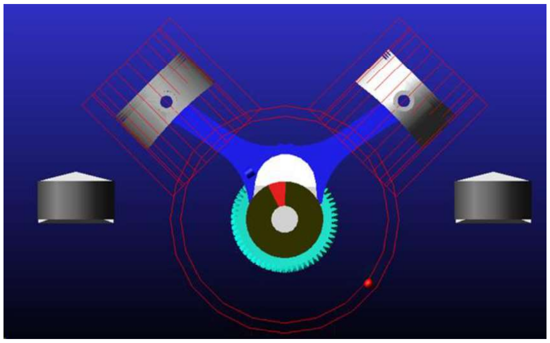


Fig. 7. 90 Degree V-Twin

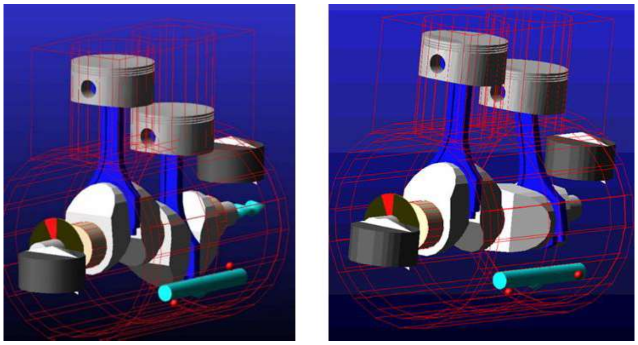


Fig. 8. 180° Firing and 270° Firing Parallel Twin.

F. *BOXER TWIN*

The boxer engine of Fig. 9 is one of the easiest configurations to model and analyse. It is essentially a parallel twin with the cylinders laid down 180 degrees from each other and firing at the exact same time. Firing at the same moment will eliminate any horizontal shaking forces and there is no vertical due to the engine lying flat in the horizontal plane. The only thing that could be concerning about this layout would be the roll moment.

G. *SIMULATION*

The largest take away from the results of this study is how dependent secondary vibration is to the engine configuration. It is tied to the specific geometry and is a lot more difficult, and expensive, to reduce. The second note to remember is that these configurations are theoretical and were balanced by hand, followed by multiple iterations to reduce the primary unbalance. There is room for further refinement to reduce shaking forces and moments by spending more time balancing each configuration and building more specific, production ready, geometry. Each configuration is the best in its own situation or goal at hand. If reducing shaking forces is the end goal, adding a 2nd order balance shaft to the current baseline configuration would be the best route, both in the vibration standpoint as well as the cost standpoint. This is done by adding the balance shaft between the Vee spinning the same direction as the crank at 2x speed. This results in reducing the shaking forces to close to zero, all while not increasing the moments. The results are shown in Tables 4 and 5 and Figs. 10 and 11.

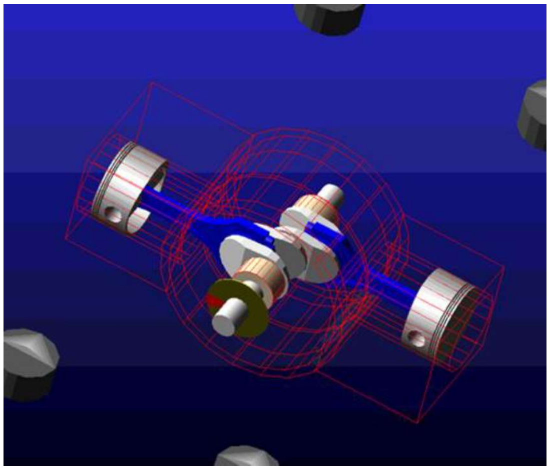
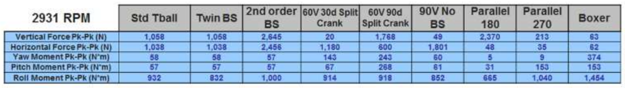


Fig. 9. Boxer Engine Configuration

Table 4. 2931 RPM Configuration Results



2931 RPM	Std Tball	Twin BS	2nd order BS	60V 30d Split Crank	60V 90d Split Crank	90V No BS	Parallel 180	Parallel 270	Boxer
Vertical Force Pk-Pk (N)	1,058	1,058	2,645	20	1,768	49	2,370	213	63
Horizontal Force Pk-Pk (N)	1,038	1,038	2,456	1,180	600	1,801	48	35	62
Yaw Moment Pk-Pk (N*m)	58	58	57	143	243	60	5	9	374
Pitch Moment Pk-Pk (N*m)	57	57	57	67	268	61	31	153	153
Roll Moment Pk-Pk (N*m)	932	832	1,000	914	918	852	665	1,040	1,454

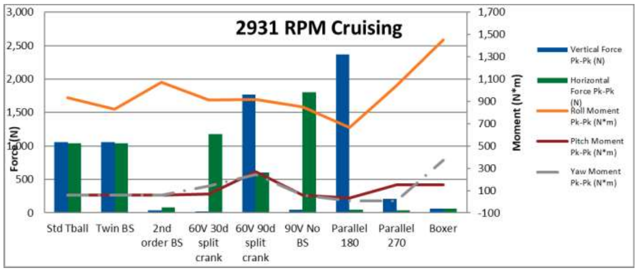
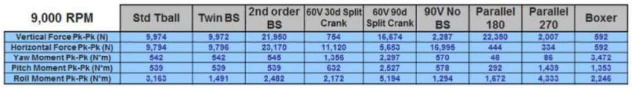


Fig. 10. 2931 RPM Configuration Results

Table 5. 9000 RPM Configuration



9,000 RPM	Std Tball	Twin BS	2nd order BS	60V 30d Split Crank	60V 90d Split Crank	90V No BS	Parallel 180	Parallel 270	Boxer
Vertical Force Pk-Pk (N)	9,974	9,972	21,950	754	16,674	2,287	22,350	2,007	592
Horizontal Force Pk-Pk (N)	9,794	9,796	23,170	11,120	5,653	16,995	444	334	592
Yaw Moment Pk-Pk (N*m)	542	542	545	1,356	2,297	570	48	86	3,472
Pitch Moment Pk-Pk (N*m)	539	539	539	632	2,527	578	292	1,439	1,353
Roll Moment Pk-Pk (N*m)	3,163	1,491	2,482	2,172	5,194	1,294	1,672	4,333	2,246

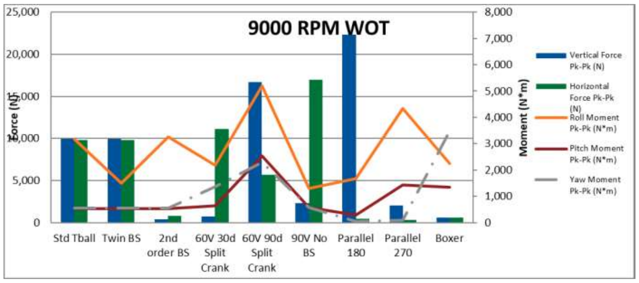


Fig. 10. 9000 RPM Configuration Results

VI. CONCLUSION

In this paper we investigated the vibrations of a twin-cylinder internal combustion engine due to engine configuration kinematics and inertial forces. Using a baseline motorcycle engine, five iterations were conducted to reduce the second order vibration. These changes include titanium pins, 2 ring pistons (changed from typical 3 ring piston), aggressive piston weight reduction, titanium connecting rods, and brushless connecting rod. The results show a noticeable reduction in the shaking forces and moments when compared to the baseline. For the first order vibrations problem, nine different configurations in combination of the angles between the connecting rods and balance shafts were investigated. The results suggest that careful engine configuration and the addition of balance shaft greatly reduces engine vibrations and consequently provides better rider experience and engine longevity.

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